

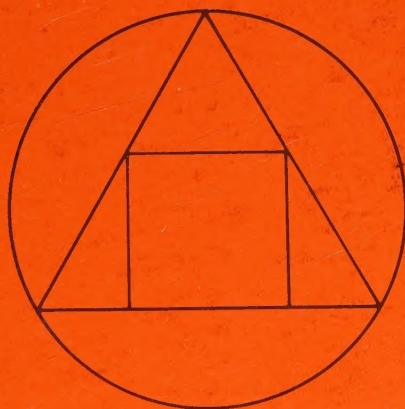
Doctor's thesis

MACHINE ELEMENTS

DIVISION

Lund Technical University

Lund, Sweden



On V-belt Drives with Special Reference
to Force Conditions, Slip, and Power Loss

B. Göran Gerbert

Lund 1973

ON V-BELT DRIVES WITH SPECIAL REFERENCE TO
FORCE CONDITIONS, SLIP, AND POWER LOSS

Av

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Tekn. lic., Gb

AKADEMISK AVHANDLING

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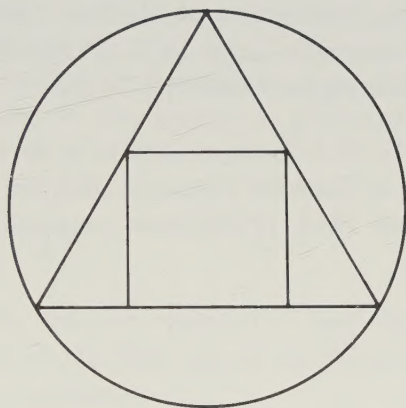
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The following papers are included in this dissertation:

1. Gerbert, B.G.: Force and slip behaviour in V-belt drives. Acta Polytechnica Scandinavica. Mech. Eng. Series No. 67. Helsinki 1972.
2. Gerbert, B.G.: Tensile stress distribution in the cord of V-belts. ASME paper No. 72-DE-D, 1972.
3. Gerbert, B.G.: Power loss and optimum tensioning of V-belt drives. Transaction of Machine Elements Division. Lund Technical University. Lund 1972. (Publication prepared in Journal of Engineering for Industry.)
4. Gerbert, B.G.: Pressure distribution and belt deformation in V-belt drives. Transaction of Machine Elements Division. Lund Technical University. Lund 1973. (Publication prepared in Journal of Engineering for Industry.)
5. Gerbert, B.G.: Scheibenspreizkräfte in Breitkeilriemengetrieben. Transaction of Machine Elements Division. Lund Technical University. Lund 1973. (Publication prepared in Antriebstechnik.)



1. Force and slip behaviour in V-belt drives

The power transmitting capacity of a V-belt drive depends on the friction between the belt and the pulley. The belt can slide both circumferentially and radially in the V-groove of the pulley. Thereby friction forces appear which are directed opposite to the sliding velocity. Only the circumferential components of the friction forces contribute to the transmitted torque.

The direction of the sliding velocity is determined by

Equilibrium conditions

Force-deformation conditions

Sliding conditions

Geometric conditions

Fig. 1 shows the loading of the cross section of the belt. The radial displacement x due to this loading is proposed to follow the equation

$$x = \frac{2p_z}{k_1} + \frac{F}{\rho k_2} + \frac{F}{k_3}$$

where k_1 , k_2 , and k_3 shall be regarded as belt constants. In this equation the influence of the direction of the friction forces on the radial displacement is considered.

The solution of the equations (which consider the variation in the radius of curvature of the belt and thus the entrance and exit regions) reveals that there exists a point where the sliding velocity $v_s = 0$ and the frictional forces are directed radially outwards i.e. the sliding angle $\gamma = 180^\circ$.

Neglect of the variation in radius of curvature and a special treatment of the point where $v_s = 0$ and $\gamma = 180^\circ$ reveal that two solutions exist at this point. The one solution corresponds to the driven pulley and the other to the driving one according to fig. 2.

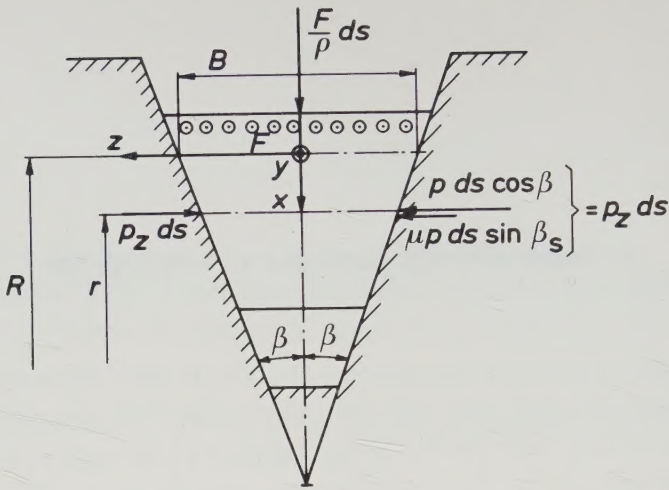
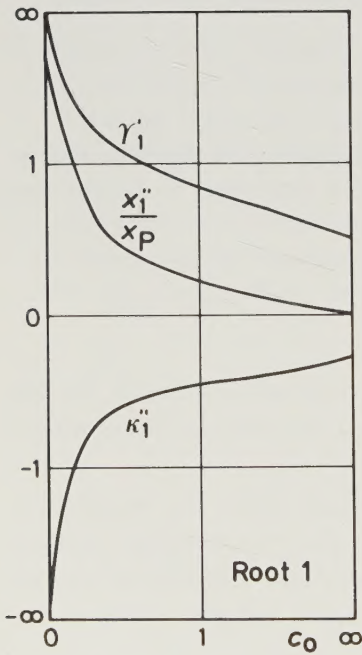
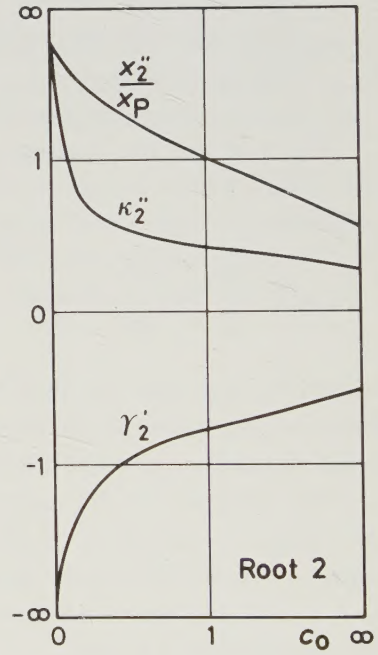


Fig. 1. Cross section of a V-belt and acting forces



a) Driving pulley



b) Driven pulley

Fig. 2. Derivatives in the point $v_s = 0$ and $\gamma = 180^\circ$ as function of the longitudinal spring constant c_0 . Coefficient of friction $\mu = 0.4$ and groove angle $\beta = 18^\circ$. κ = relative force (force variation), x = radial displacement, and γ = sliding angle.

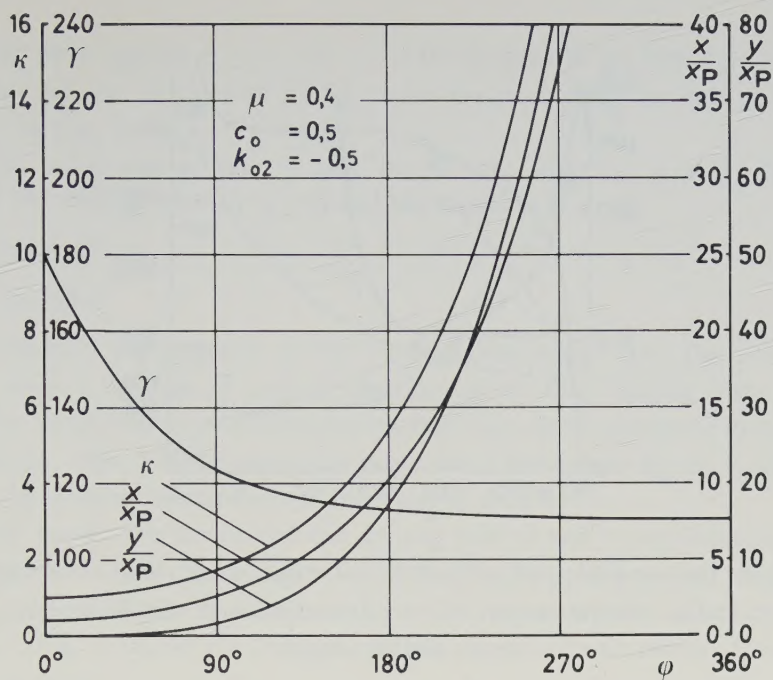


Fig. 3. Solution for a driven pulley. κ = relative force, x = radial displacement, and γ = sliding angle.

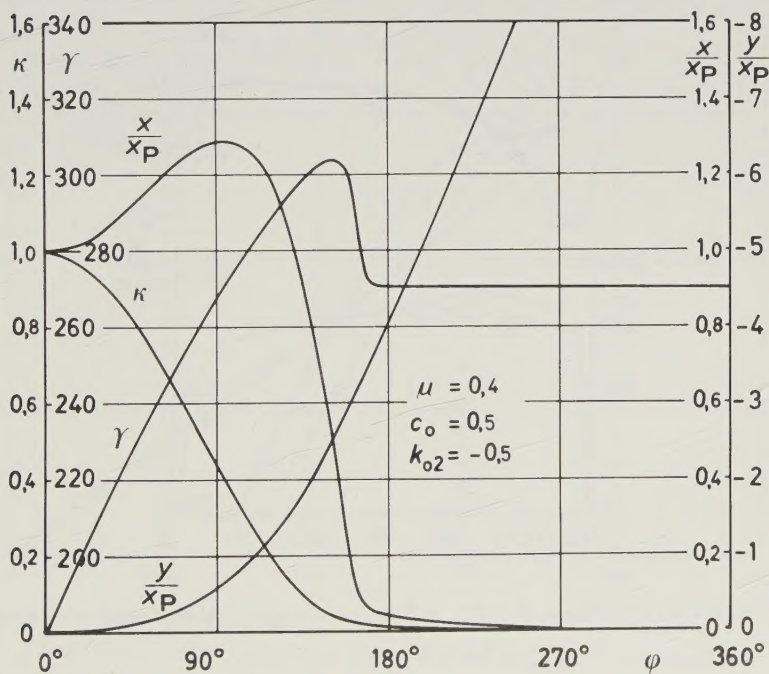


Fig. 4. Solution for a driving pulley

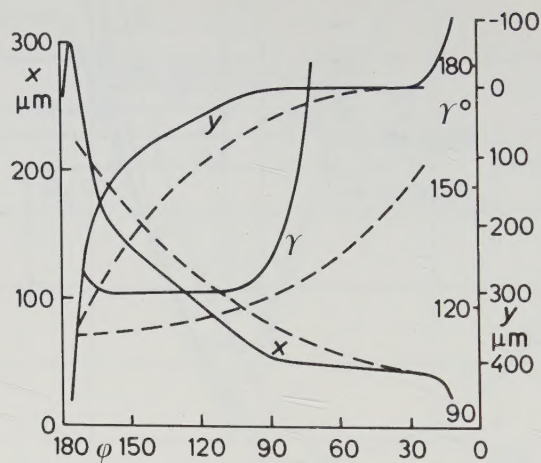


Fig. 5. Driven pulley. $F_2/F_1=3.9$. — experiments and ----- theory. x = radial displacement, y = circumferential displacement, and γ = sliding angle.

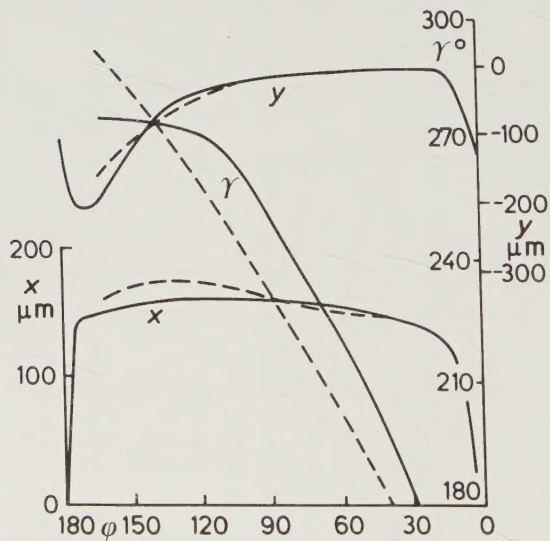


Fig. 6. Driving pulley. $F_2/F_1 = 3.9$. — experiments and ----- theory.

The point where $v_s = 0$ and $\gamma = 180^\circ$ is used as an initial point. The use of the derivatives in fig. 2 in the start gives the solutions shown in figs. 3 and 4.

The solutions in figs. 3 and 4 are only valid for $\varphi \geq 0$. For $\varphi \leq 0$ the sliding velocity $v_s = 0$ and the solution is

$$x/x_p = 1$$

$$\gamma = 180^\circ$$

This solution corresponds to the "non-sliding zone" and the solutions shown in figs. 3 and 4 correspond to the "sliding zone".

The radial and circumferential motion have been measured in experiments. Figs. 5 and 6 show the result for a force ratio $F_2/F_1 = 3.9$. Even the corresponding theoretical result is shown.

The results for the driven and driving pulleys can be combined to a complete V-belt drive. Quantities such as slip, efficiency, axial force, and belt forces in a drive with locked centre distance have been treated. Fig. 7 shows the theoretical and experimental results of slip investigations.

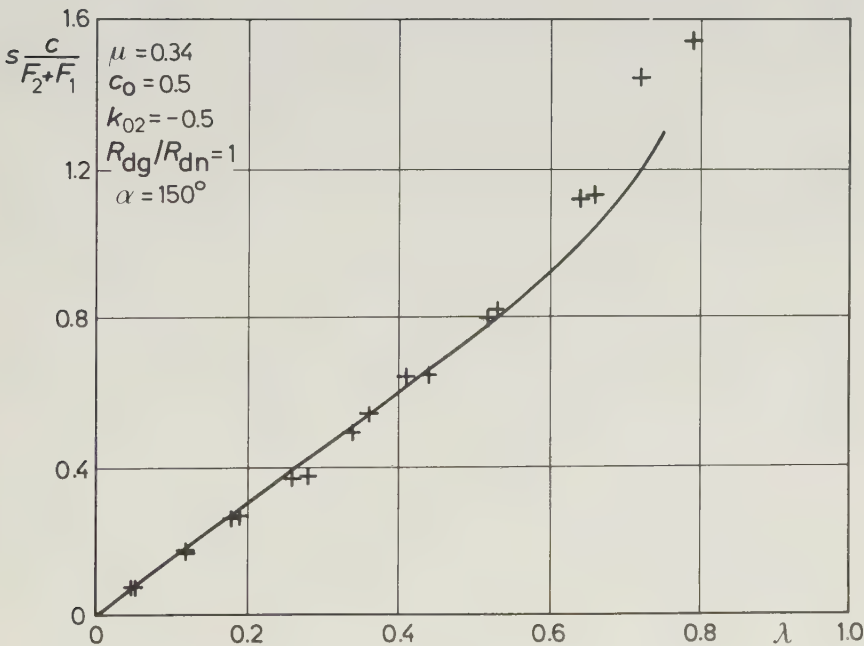


Fig. 7. Theoretical slip factor as function of $\lambda = (F_2 - F_1) / (F_2 + F_1)$ and experimental results

2. Tensile stress distribution in the cord of V-belts

Almost the whole belt force is carried by the cord so the friction forces on the sides of the belt must be transmitted through the cover and the rubber and then be taken up by the cord. Due to the construction of the belt this transmission must give an irregular force distribution in the cord. Fig. 8 shows the variation in strain difference between the side and the middle of a V-belt. Both theoretical and experimental results are shown. The variation of the friction forces along the belt are calculated according to the theory in part 1.

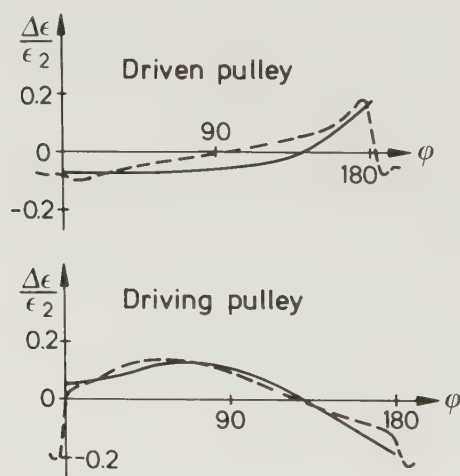


Fig. 8. Theoretical (—) and experimental (---) results for strain difference in the cord between the side and the middle of the belt.

$F_2/F_1=6$. ϵ_2 =mean strain in the tight part.

3. Power loss and optimum tensioning of V-belt drives

The power loss can be divided into two parts viz. losses due to external friction (sliding between belt and pulley) and internal friction (sliding between molecules-hysteresis). The former losses are treated theoretically and the latter ones experimentally.

To be able to calculate the losses caused by external friction the regions where the belt enters and leaves the pulley must be included in the analysis. This means that the theory in part 1 must be extended to consider the bending moment of the belt and the variation of the sliding angle across the belt. The difference between the results from this extended theory and the results from part 1 gives the contribution to the power loss in the entrance and the exit regions and according to fig. 9 these losses can be separated from other external friction losses.

Losses due to internal friction depend on the molecular properties of the type of rubber, cord, and cover which are used in the belt. The losses can be divided into bending losses and compression losses, which are measured experimentally. Internal friction losses due to the elongation of the belt can be neglected.

All contributions can be put together to the total power loss which is shown in fig. 10. This figure even shows experimental results. If the initial tension of the drive is varied it is possible to find an optimum working point where a certain combination of initial tension and transmitted torque gives a maximum efficiency.

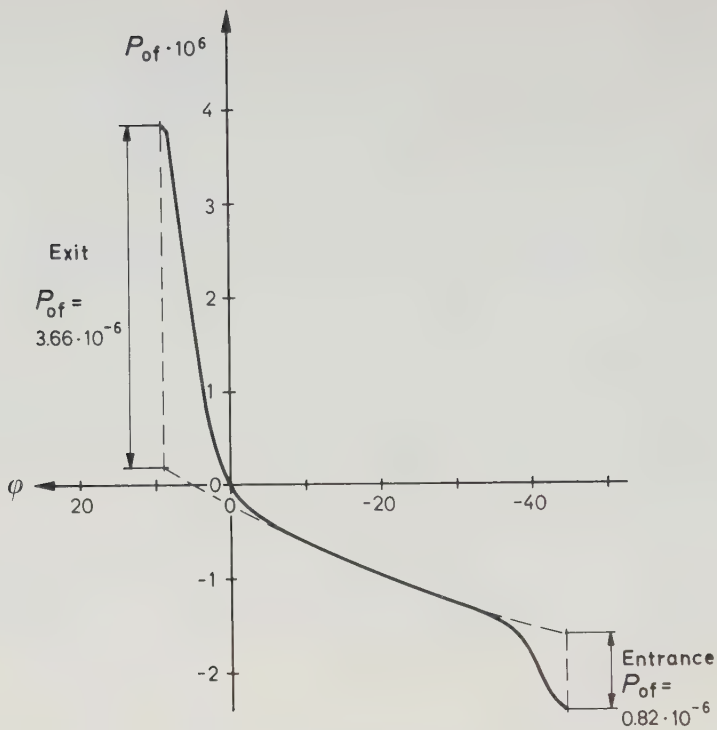


Fig. 9. Power loss as function of the angular coordinate. Driven pulley. The dotted line is the power loss if the entrance and the exit regions are neglected.

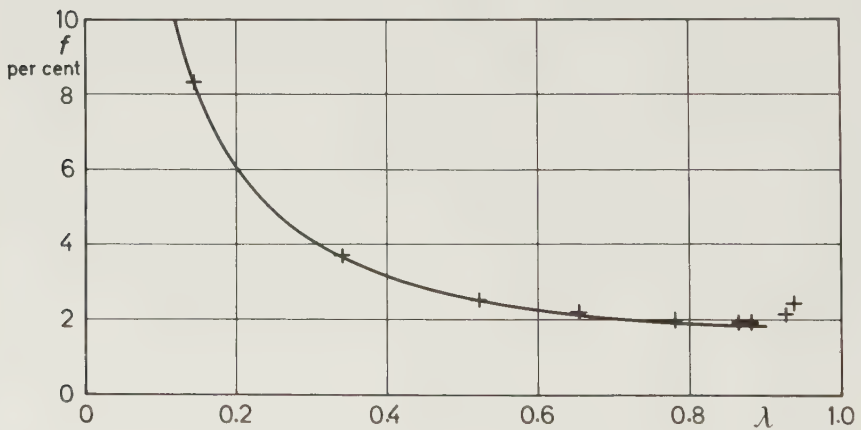


Fig. 10. Theoretical power loss factor and experimental results for a drive with two pulleys of equal size. Initial tension 200 N.

4. Pressure distribution and belt deformation in V-belt drives

A radial component of the cord force forces the belt into the groove of the pulley and a pressure appears on the side of the belt. The distribution of this side pressure depends on the distribution of cord force, the friction, the shape of the groove, and the construction of the belt. The pressure is calculated by means of finite element analysis and a result is shown in fig. 11.

Fig. 11 even shows the radial displacement. In part 1 a formula for the radial displacement was proposed and the belt constants k_1 , k_2 , and k_3 were determined by experiments. These led to $k_1/E = 2.32$ and $k_2/k_1 = -0.57$. The theoretical calculation of these constants led to $k_1/E = 2.41$ and $k_2/k_1 = -0.96$. E is here the modulus of elasticity of the rubber.

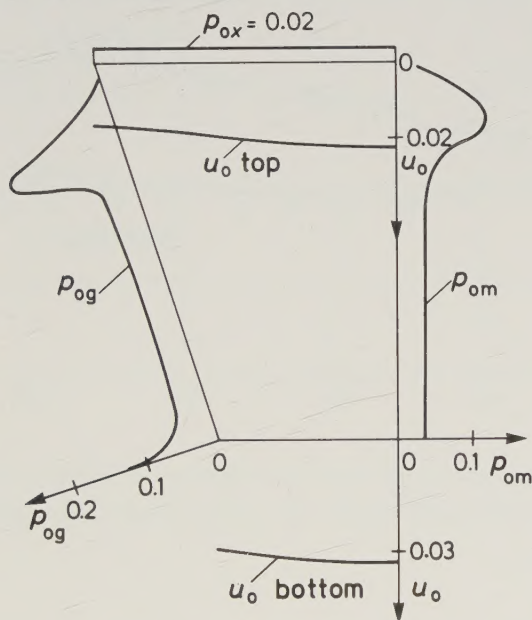


Fig. 11. Pressure distribution and radial deformation. The figure shows half the belt. Compressive modulus of elasticity of the cord is 25 times that of the rubber.

5. Axial force in wide belt adjustable speed devices

There are different ways to stretch V-belt drives e.g. by means of weights, locked centre distance. In adjustable speed devices very often one of the pulleys is springloaded and in such a case the axial force determines the initial tension and the traction properties of the drive, which is shown in fig. 12.

The wrap angle (angle of contact) influences the axial force. If large torques are transmitted the wrap angle decreases due to the bending stiffness of the belt and if the springloaded pulley is mounted on the driven shaft a considerable decrease in traction capacity is obtained.

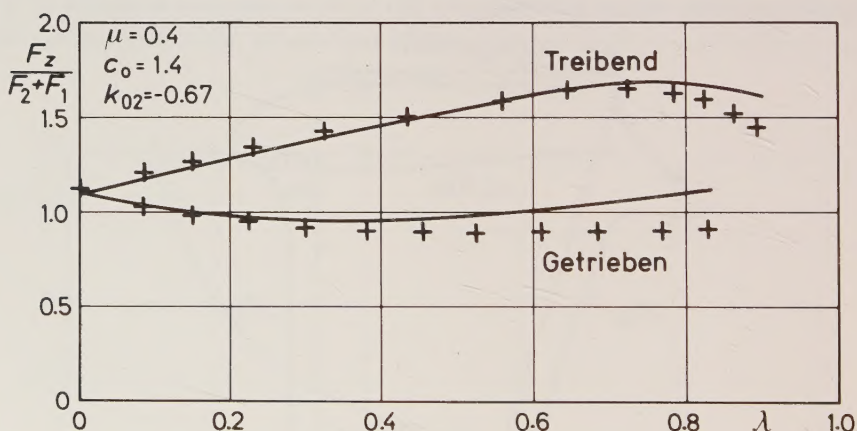


Fig. 12. Theoretical axial force as function of $\lambda = (F_2 - F_1) / (F_2 + F_1)$ and experimental results. Both the pulleys have the same pitch radius.

